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MEASUREMENT OF WHEEL/RAIL FORCES AT THE WASHINGTON METROPOLITAN AREA TRANSIT AUTHORITY

Volume I. Analysis Report

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JULY 1980

INTERIM REPORT

Prepared for

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Office of Rail and Construction Technology
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16. Abstract

Under the direction of the Urban Mass Transportation Authority (UMTA) measurements of wheel/rail forces were made in August 1979 by the Transportation Systems Center (TSC) with the assistance of Battelle Columbus Laboratories to determine the causes of excessive wheel/rail wear experienced by the Washington Metropolitan Area Transit Authority (WMATA) Metrorail System. In addition to measuring the absolute magnitude of the wheel/rail forces, it was the intent to compare alternative methods for relieving wheel/rail wear.

For tight gage the average flange force between the leading outer wheel and the high rail of an 800-foot radius curve was 9400 pounds, unworn cylindrical profile; 6300 pounds, unworn tapered profile; 7900 pounds, worn cylindrical profile. For widened gage the average flange force was 6300 pounds, unworn cylindrical profile; 5500 pounds, unworn tapered profile. On the basis of these results it was recommended that cylindrical wheels be replaced by tapered wheels and tight gage curves be widened to standard gage.

This report consists of two volumes:

- Volume I, The Analysis Report, analyzes the data and presents conclusions and recommendations.
- Volume II, The Test Report, describes the wayside sites and instrumentation and presents the wheel/rail load data from the test runs in a tabular format.

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PREFACE

This report describes measurements made of wheel/rail forces on the Washington Metropolitan Area Transit Authority (WMATA) Metrorail System in order to determine causes of excessive wheel/rail wear and make recommendations for its alleviation.

The test program was sponsored by the U.S. Department of Transportation (DOT), Urban Mass Transportation Administration (UMTA), through the Office of Rail and Construction Technology of the Office of Technology Development and Deployment. The work was performed by the Transportation Systems Center (TSC) with the assistance of the Battelle Columbus Laboratories.

This measurement program involved the contributions of many people. In particular, the authors wish to thank:

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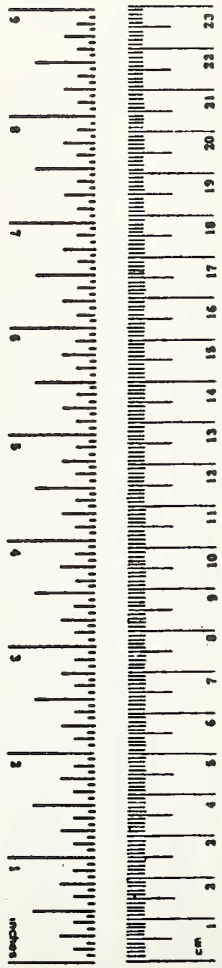
David Gaul Ed Green	WMATA, Office of Equipment Design
Adelbert Lavery	TSC, Transit Systems Branch
Lennart Long Melvin Yaffee Robert Rudis Charles Dunn	TSC, Equipment and Controls Branch

METRIC CONVERSION FACTORS

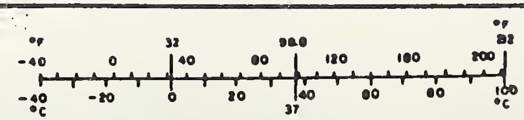
Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C



Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F



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LIST OF ABBREVIATIONS AND SYMBOLS

W/R	wheel/rail
H	high rail
V	vertical force
L	lateral force
n	number of samples
L/V	ratio of lateral to vertical wheel/rail forces
μ	coefficient of friction
f_L	creep coefficient in the lateral direction
f_T	creep coefficient in the tangential direction
ℓ	half of track gage
α	wheel conicity
r_o	wheel radius of undisplaced wheelset
W	wheel load
K_y	effective lateral stiffness of flexible truck
K_ψ	effective yaw stiffness of flexible truck

1.0 INTRODUCTION

1.1 BACKGROUND

The Washington Metropolitan Area Transit Authority (WMATA) has noted excessive wheel and rail wear during the first three years of operation. A study was conducted by DeLeuw, Cather and Company to determine the magnitude of the wear problem. Findings published in a report in March 1979 indicated that accelerated rail wear was limited to curves of less than a 1000 ft radius. These curves represented less than 6 percent of the Washington Metro operating mileage at that time. A comprehensive wear survey of 40 percent of all such curves indicated that maximum wear existed at the outbound track of curve #1 at Washington National Airport (see Figure 1).

As a result of the concern over excessive wear and its implication for all transit properties the Urban Mass Transportation Authority (UMTA) funded the Transportation Systems Center (TSC) and WMATA to make measurements in order to determine the causes of the problem and make recommendations for its alleviation. In August of 1979, TSC with the assistance of Battelle Columbus Laboratories measured wheel/rail forces at the Washington National Site and the Brentwood Site. This report describes the results of that effort. Volume II of this report records the instrumentation, test descriptions and results of the wheel/rail force data.

Volume I, the Analysis Report, reduces the test report data and evaluates that data in terms of absolute wheel/rail and flange force values for the various test conditions. As a result, it establishes relative comparisons of various methods for reducing the wear problem.

1.2 PURPOSE

The purpose of the WMATA Wheel/Rail Force Measurements was to determine the absolute magnitude of the wheel/rail forces as they relate to wheel and rail wear, and to compare alternative methods for relieving wheel/rail wear as follows:

FIGURE 1. WHEEL/RAIL FORCE MEASUREMENT SITE AT WASHINGTON NATIONAL AIRPORT

a. Use of tapered wheels instead of cylindrical wheels

b. Widening the gage on curves

c. Use of restraining rail on curves

d. Use of lubrication on curves

1.3 TEST NOTATION

The wheel/rail forces were measured at four separate locations on the curve at Washington National Airport site (see Figure 2). Lateral force measurements were designated L1, at location one, L2, at location two, etc. Vertical force measurements were designated V1, at location one, V2, at location two, etc. Forces on the outside or high side of the curve, directed inward against the wheel, were indicated HI and positive in value. Forces on the inside or low side of the curve, directed inward against the wheel, were indicated LO and positive in value.

The axles of the test consist were numbered in sequence starting with the leading car in the direction of travel (see Figure 3). Odd-numbered axles were the leading axles of each truck and even-numbered axles were trailing. The test consist was looped at the end of selected days of testing allowing the car equipped with cylindrical wheels to lead on one day and the car equipped with tapered wheels to lead on another day of runs.

The test runs were noted by the date they were made. (Example: Run 15-2 was the second run on the 15th of August.)

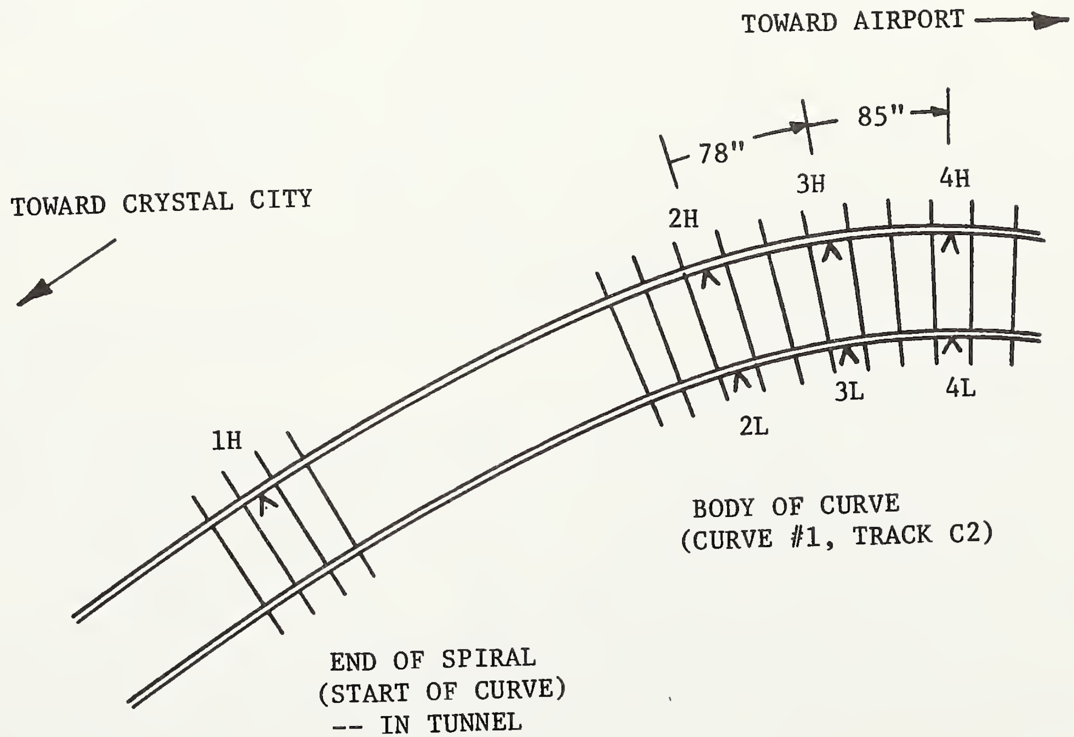

FIGURE 2. PLAN VIEW OF NATIONAL AIRPORT TEST SITE

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15. Supplementary Notes

"Volume II: Test Report" (UMTA-MA-06-0025-80-7), describes the wayside sites and instrumentation and presents the wheel/rail load data from the test runs in a tabular format.

16. Abstract

Under the direction of the Urban Mass Transportation Administration (UMTA), measurements of wheel/rail forces were made in August 1979 by the Transportation Systems Center (TSC) with the assistance of Battelle Columbus Laboratories to determine the causes of excessive wheel/rail wear experiences by the Washington Metropolitan Area Transit Authority (WMATA) Metrorail System during its first three years of operation. In addition to measuring the absolute magnitude of the wheel/rail forces, it was the intent to compare alternative methods for relieving wheel/rail wear at WMATA and other transit properties. Measurements of the wheel/rail forces were made at the Washington National Airport Test Site and the Brentwood Shop Test Site. This report describes the results of that effort.

The study found that for tight gage, the average flange force between the leasing outer wheel and the high rail of an 800-foot radius curve was 9400 pounds, unworn cylindrical profile; 6300 pounds, unworn tapered profile; and 7900 pounds, worn cylindrical profile. For widened gage, the average flange force was 6300 pounds, unworn cylindrical profile and 5500 pounds, unworn tapered profile. On the basis of these results, it was recommended that cylindrical wheels be replaced by tapered wheels and tight gage curves be widened to standard gage.

This report consists of two volumes. This report, Volume I, reduces the test report data and evaluates that data in terms of absolute wheel/rail and flange force values for the various test conditions. As a result, it establishes relative comparisons of various methods for reducing the wear problem.

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Rail Dynamics; Rapid Rail Transit; Vehicle Dynamics; Washington Metropolitan Area Transit Authority; Wear Reduction; Wheel/Rail Forces; Wheel/Rail Wear; WMATA	Available to the public through the National Technical Information Service, Springfield, Virginia 22161.

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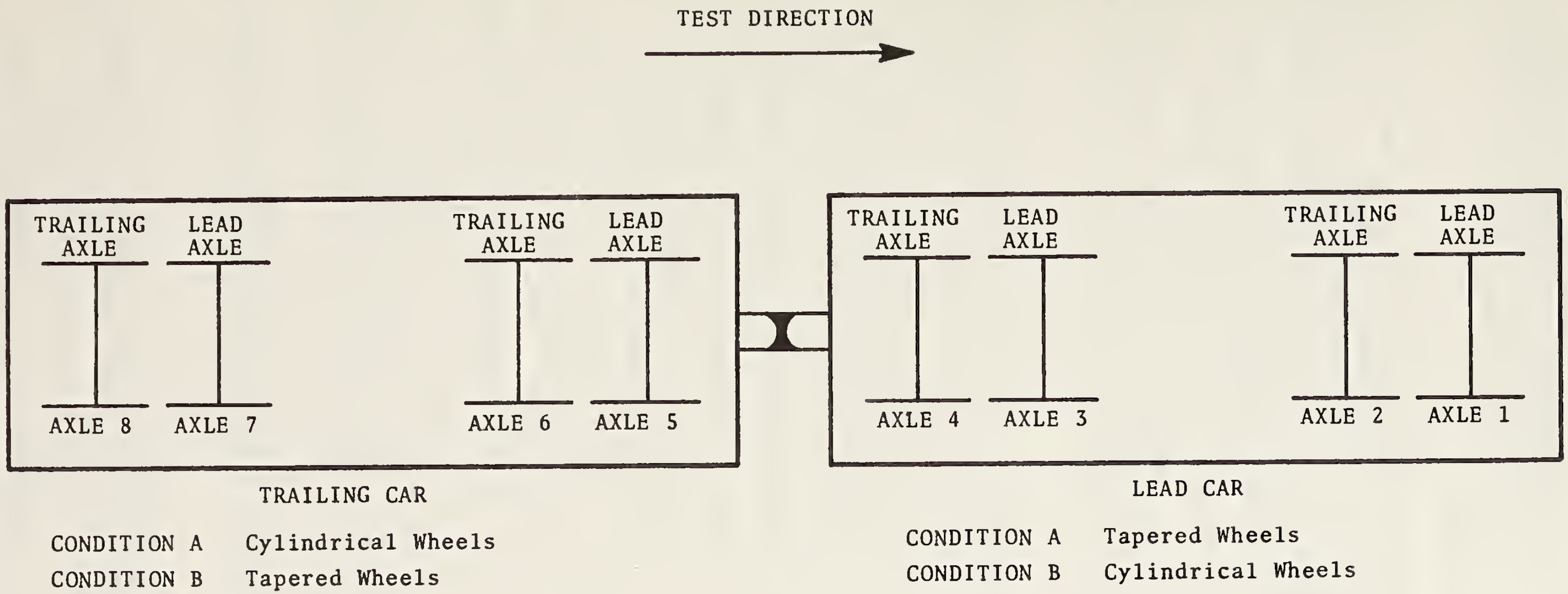


FIGURE 3. DIAGRAM OF THE TEST CONSIST INDICATING LOCATION OF AXLES IN RELATION TO TEST DIRECTION

2.0 ANALYSIS OF WHEEL/RAIL FORCES AND FLANGE FORCES

2.1 TYPICAL REVENUE SERVICE CONDITIONS

In order to establish a baseline for recording and evaluating data under normal operating conditions, measurements of train loads due to revenue service were recorded. Measurements of these 6- or 8-car consists were made at both the National and Brentwood sites. At the National Airport test site, revenue trains approached the instrumented section at 20-25 mph, slowing to 5-10 mph by the time they passed the test site, and sometimes braking to a stop on the instrumented section. At the Brentwood Shop test site, revenue trains passed at the programmed speed of 53 mph.

Some of the revenue data recorded at the sites are shown in Volume II. Using all the revenue data for the National runs an analysis of force levels was done. Throughout the curve, the lead axles produce lateral wheel/rail (W/R) forces that are much greater than the trailing axle forces. The W/R forces generated at each location are fairly uniform. The greatest variation occurred at location 3 for the lead axle with a mean lateral high rail W/R force of 3800 lbs and a standard deviation of 1100 lbs (28 percent of the mean). The mean lateral high rail W/R force at locations 2 and 4 was 3100 lbs and 3400 lbs with equal standard deviations of 600 lbs. Because of the wide variety of operating conditions during these revenue runs, these results indicate that the effect of velocity variation during curving is not major. For the lead axles, the maximum force tends to occur at location 3. Averaging these mean high rail lead axle forces over the curve at locations 2, 3, and 4 leads to an overall curving W/R force of 3400 lbs (see Table 1). In order to further evaluate the force levels while traversing the curve, the mean W/R force versus the location in the curve is graphed in Figure 4. The forces are graphed in terms of the 95 percent confidence interval about the mean. The lead axle forces are consistently much higher than the trailing axle forces; both tend to peak at location 3.

The wheel/rail force recorded in these tests is the net load which is composed of the creep force and the flange force. The flange forces are calculated in the manner outlined in the Appendix. These flange forces and the frictional forces generated by flange friction are important factors in the

TABLE 1. SUMMARY OF THE MEAN RAIL FORCES AND MEAN CALCULATED FLANGE FORCES FOR ALL REVENUE SERVICE RUNS MEASURED AT THE WASHINGTON NATIONAL AIRPORT SITE FOR 204 ODD- AND 204-EVEN NUMBERED AXLES

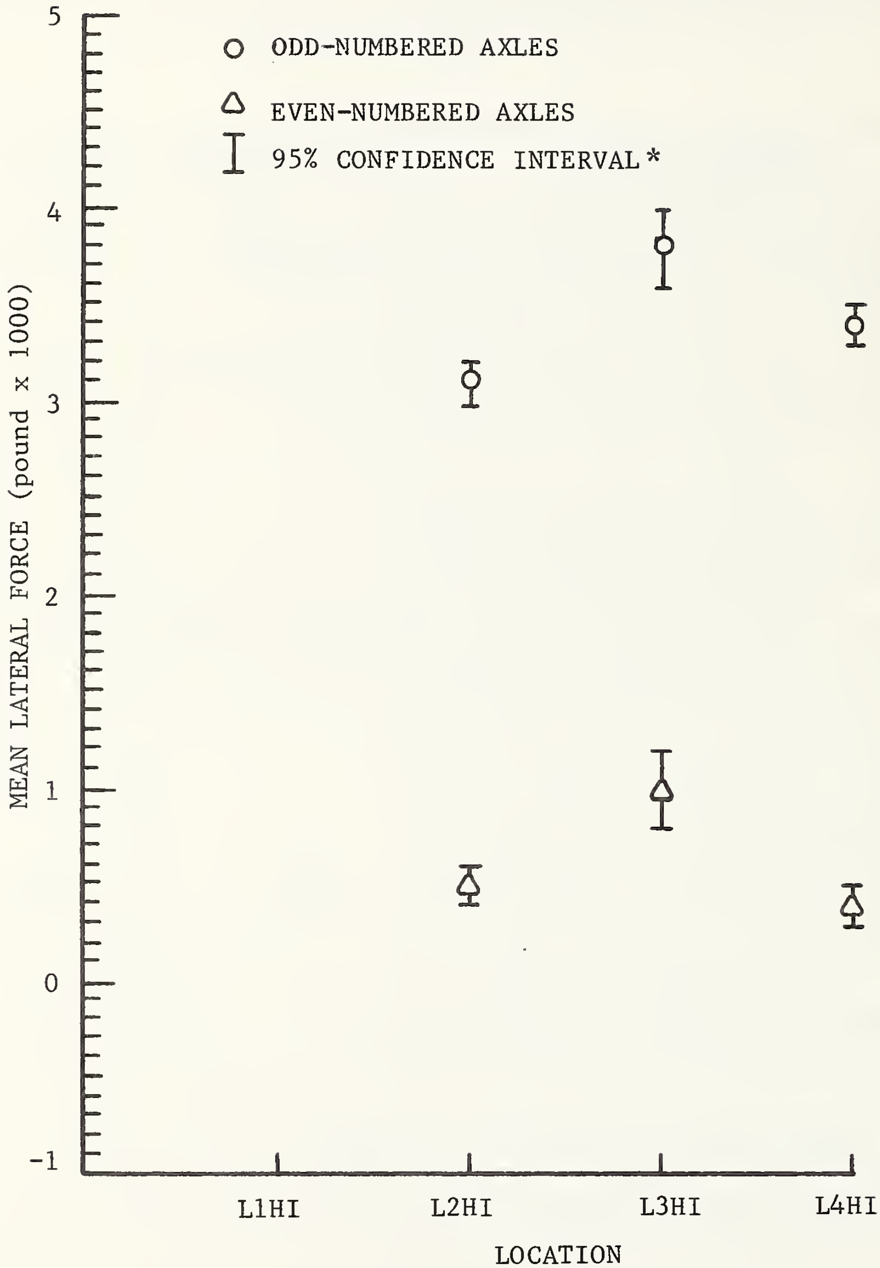
RAIL FORCE CIRCUIT	MEAN RAIL FORCE* (IN POUND UNITS)		MEAN CALCULATED FLANGE FORCE* (IN POUND UNITS)	
	LEAD AXLES	TRAILING AXLES	LEAD AXLES	TRAILING AXLES
V2HI	10,100 (1200)	9700 (1500)	-	-
L2HI	3100 (600)	500 (700)	6800 (900)	2300 (1100)
V2LO	9100 (1100)	9700 (900)	-	-
L2LO	3600 (700)	1800 (600)	-	-
V3HI	11,800 (1700)	10,100 (1700)	-	-
L3HI	3800 (1100)	1000 (1300)	9000 (2100)	3400 (2600)
V3LO	10,500 (1100)	9500 (1800)	-	-
L3LO	4500 (900)	1900 (1300)	-	-
V4HI	10,800 (1200)	10,100 (1200)	-	-
L4HI	3400 (600)	400 (300)	n/a**	n/a**
V4LO	n/a	n/a	-	-
L4LO	3000 (700)	600 (400)	-	-

MEAN HIGH RAIL LATERAL FORCE OVER CURVE 3433 LBS.

MEAN LOW RAIL LATERAL FORCE OVER CURVE 3700 LBS.

*The numbers in parenthesis are sample standard deviations with n-1 weighting.

**The flange forces at L4HI were not available because the V4LO was not operational during the revenue service runs.



*The true mean value of force lies within the range shown with a 95% probability

FIGURE 4. GRAPH OF THE MEAN LATERAL FORCE (IN POUNDS) GENERATED ON THE HIGH RAIL BY WHEELS OF WMATA REVENUE CONSISTS AT THE NATIONAL SITE

wear conditions in the entire WMATA system. The creep coefficients used in the calculation of the flange force are dependent on the ratio of the vertical wheel loads. The flange forces can be substantially greater than the W/R forces. In Table 1, which is based on all the revenue service runs at the National site, the mean flange force for the lead axles is estimated to be more than twice the value of the mean W/R force at the rail force circuit, L2HI, (6800 lbs vs 3100 lbs), while at L3HI the ratio of flange force to W/R force is 2.4 (9000 lbs vs 3800 lbs).

The maximum lateral force developed during curve traversal by the revenue consists is also of importance. The maximum W/R forces recorded at locations 2, 3, and 4 on the high rail are shown in Table 2 for selected revenue runs. The maximum forces recorded at location 3 are always higher than the maximums recorded at the adjacent locations 2 and 4. The largest maximum force in this sample is 5900 lbs at L3HI.

A similar series of measurements were made for the revenue service runs at the Brentwood sites. The maximum lateral loads developed at this site are recorded in Table 3 for the runs on August 20. The maximum force of 2500 lbs is less than half of the maximums recorded at the National site listed in Table 2. In general, the lateral forces were much lower than the comparable forces at the National site. The lower forces at Brentwood are due to the increased curve radius (1527 feet at Brentwood vs 800 feet at National) and the presence of rain during the Brentwood measurements. In Table 4, the mean forces at location 2 for all the Brentwood revenue runs are tabulated and the maximum lateral W/R force is 1300 lbs. The flange force for this location is similarly low and is calculated to be 1600 lbs.

2.2 CONTROLLED TESTS AT THE WASHINGTON NATIONAL SITE

A 2-car test consist, one car equipped with standard cylindrical wheels and the other with tapered* wheels of 1:20 profile, was operated at this site. Some tests were run with the cylindrical-wheeled car in the lead, others with the tapered-wheeled car in the lead.

*Tapered is a word used throughout this report to describe a conically shaped wheel of British Rail Profile.

TABLE 2. SUMMARY OF THE MAXIMUM LATERAL RAIL FORCES PRODUCED BY REVENUE CONSISTS AT THE NATIONAL SITE

	LATERAL RAIL FORCE (IN POUND UNITS)		
RUN NUMBER	L2HI	L3HI	L4HI
15-17	3800	4900	4300
15-18	4200	5900	4400
15-19	3900	4800	4300
15-24	3800	5600	4800
15-25	3500	4600	4200
15-26	3500	4700	4500
15-27	4500	5000	4000
16-12	4400	5600	4800
16-15	3900	5000	4500
16-16	4000	4900	3600
16-17	4600	5000	4300
16-19	4200	5400	4300
16-21	4700	5800	4400
16-23	4300	4800	4200
16-24	4200	5400	4600

TABLE 3. SUMMARY OF THE MAXIMUM RECORDED LATERAL RAIL FORCES PRODUCED BY WHEELS OF REVENUE CONSISTS AT THE BRENTWOOD SITE ON AUGUST 20th, 1979

RUN NUMBER	MAXIMUM LATERAL RAIL FORCE (IN POUND UNITS)
20-1	1400
20-2	2200
20-3	2200
20-4	2200
20-5	1700
20-6	2500
20-7	2300
20-8	1400
20-9	2300

TABLE 4. SUMMARY OF THE MEAN RAIL FORCES AND MEAN CALCULATED FLANGE FORCES FOR ALL REVENUE SERVICE RUNS MEASURED AT THE BRENTWOOD SITE FOR 80 ODD- AND 80 EVEN-NUMBERED AXLES

RAIL FORCE CIRCUIT	MEAN RAIL FORCE* (IN POUND UNITS)		MEAN CALCULATED FLANGE FORCE* (IN POUND UNITS)	
	LEAD AXLES	TRAILING AXLES	LEAD AXLES	TRAILING AXLES
V2HI	9200 (800)	9500 (600)	-	-
L2HI	200 (1200)	300 (600)	1600 (1800)	1000 (800)
V2LO	9300 (2700)	8600 (700)	-	-
L2LO	1300 (500)	600 (600)	-	-

*The number in parenthesis are sample standard deviations with n-1 weighting.

Test conditions varied: Speeds ranged from 5 to 40 mph; operating conditions included train acceleration, coasting and braking through the site; rail surface conditions were dry and lubricated; track gage was the standard WMATA tight gage or a widened gage.

2.2.1 Dry Conditions

a) Coasting

Runs under nominal conditions of standard gage showed a typical pattern of the highest lateral wheel loads recorded at the lead axle, while substantially lower lateral loads were recorded at the trailing axle of a truck. In general, lead outer wheel lateral loads were found to be higher with the cylindrical wheels than with the tapered wheels. Some of these trends may be seen from the load variation as the axles pass one fixed location. The lateral force variations as the test consists pass location L2HI for runs 17-2 and 18-1 are shown in Figures 5 and 6. For run 17-2, at 44 mph, the cylindrical-wheeled car is in front and the lead cylindrical wheels (axles 1,3) produce higher lateral forces than the lead tapered wheels (axles 5,7). The maximum lateral cylindrical wheel load is 4200 lbs (axle 1) while the tapered wheel load is almost half that amount (2200-2300 lbs, axles 5,7). It is also shown in Figure 5 that the trailing axles (2,4,6,8) have a relatively low lateral force on the high rail of less than 1000 lbs for both types of wheels. It is of interest to examine the same situation at location L2HI for the case where the tapered-wheeled car leads the consist to see if the foregoing trends change. As can be seen from Figure 6, for run 18-1 at 42 mph, the cylindrical wheel loads remain higher than the tapered wheel loads and the maximum values (4200 lbs cylindrical and 2900 lbs tapered) are similar to the loads from the previous configuration. In general, the wheel profile governs the load level produced rather than the position of the car in the consist.

Maximum values of lateral wheel loads were 8800 lbs (L/V = .69) with cylindrical profiles and 5400 lbs (L/V = .41) with tapered profiles were recorded during the tests for the lead axles on the high rail. In general, lead outer wheel lateral loads with cylindrical profiles ranged consistently up to 6000 lbs (L/V up to .5), while few lateral loads with tapered profiles exceeded 5000 lbs (L/V = .4).

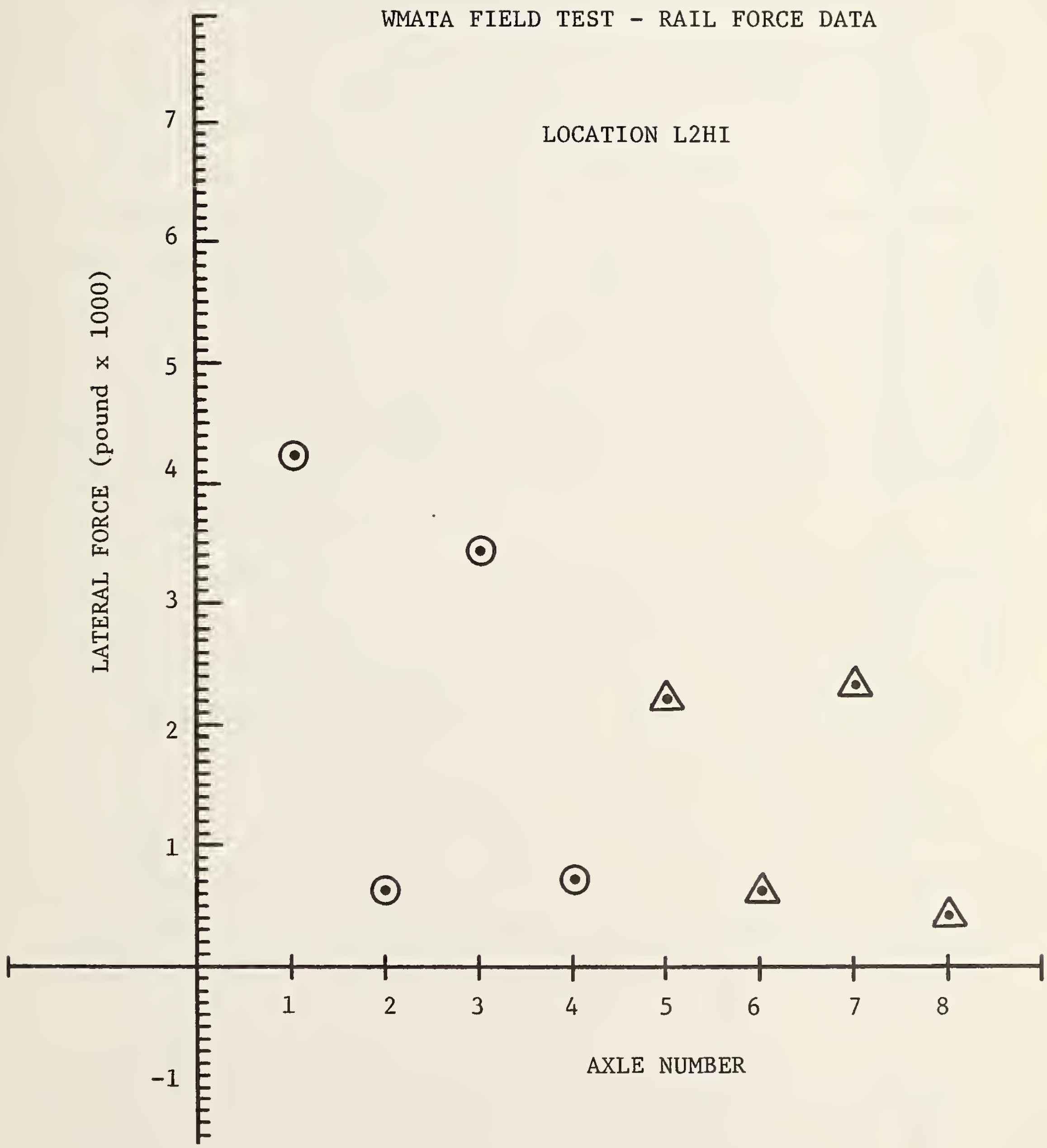


FIGURE 5. LATERAL FORCE VS AXLE NUMBER FOR AN INDIVIDUAL TEST CONSIST RUN AT NATIONAL AIRPORT (CYLINDRICAL WHEELS - AXLES 1-4, TAPERED WHEELS - AXLES 5-8, LOCATION L2HI RUN 17-2).

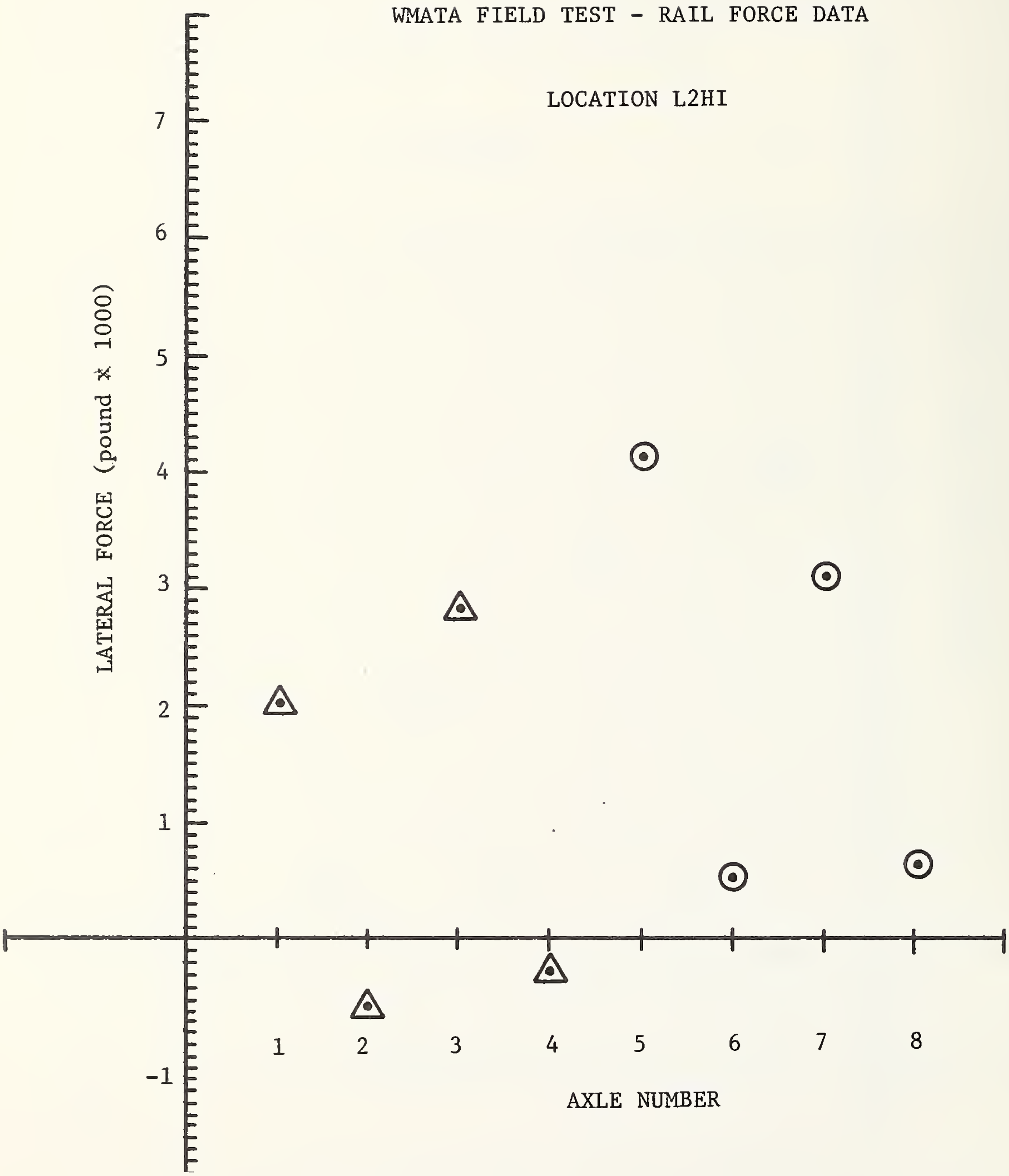


FIGURE 6. LATERAL FORCE VS AXLE NUMBER FOR AN INDIVIDUAL TEST CONSIST RUN AT NATIONAL AIRPORT (TAPERED WHEELS - AXLES 1-4, CYLINDRICAL WHEELS - AXLES 5-8, LOCATION L2HI, RUN 18-1).

Velocity variation from run to run does not appear to produce a significant trend, although the forces at location 3 show more sensitivity to velocity than those at location 2. Consider Figures 7 and 8, at locations 2 and 3, respectively, for the lead axles 1 and 5 under coast conditions on dry rail. For about half of these runs, axle 1 had cylindrical wheels; and for the other half, axle 1 had tapered wheels. For the L2HI position the lead axle cylindrical wheel loads shown in Figure 7 tend to cluster around 4100 lbs and the tapered wheel loads cluster around 2300 lbs. For the L3HI location shown in Figure 8, the cylindrical and tapered wheel loads show much more scatter with velocity and it is difficult to estimate an "average" or "clustered" value. However, these forces are about 50 percent higher than those at location L2HI (and L4HI). This trend towards higher force levels at location 3 (as in the revenue service test results) and greater sensitivity to consist velocity may imply that the trucks traverse this location differently from other locations, leading to the excessive wear condition.

The results for these runs on dry rail under various operating conditions along with a comparison of the revenue service runs are listed in Table 5, with the standard deviation (n-1 weighting) listed in the parentheses. For the coasting runs, tapered wheel profiles result in W/R forces at least 37 percent less than those associated with the cylindrical wheel profile. The advantage of the tapered wheel is greatest at location L3 where a 47 percent reduction over the cylindrical wheel is found. On the average for the coasting runs, the revenue service runs with worn cylindrical wheels indicate W/R forces about 28 percent less than the newly machined cylindrical wheels and 27 percent greater than newly machined tapered wheels. At most locations the coasting runs for both cylindrical and tapered wheels produced mean lateral W/R forces less than the forces from the accelerating runs but more than from the braking runs. This trend is not followed at L3HI where the cylindrical wheels for the coasting condition produced a higher mean force (6460 lbs) than for either the accelerating or braking runs. Further discussion of the effects of operating mode is found in Section 2.2.1 b.

The flange forces involved as the trains move through the curve may be estimated from the information in the Appendix and Table 6. The calculation of the creep coefficients are dependent on the ratio of the vertical wheel loads. Since the V4-LO recording channel was inoperative (see Volume II

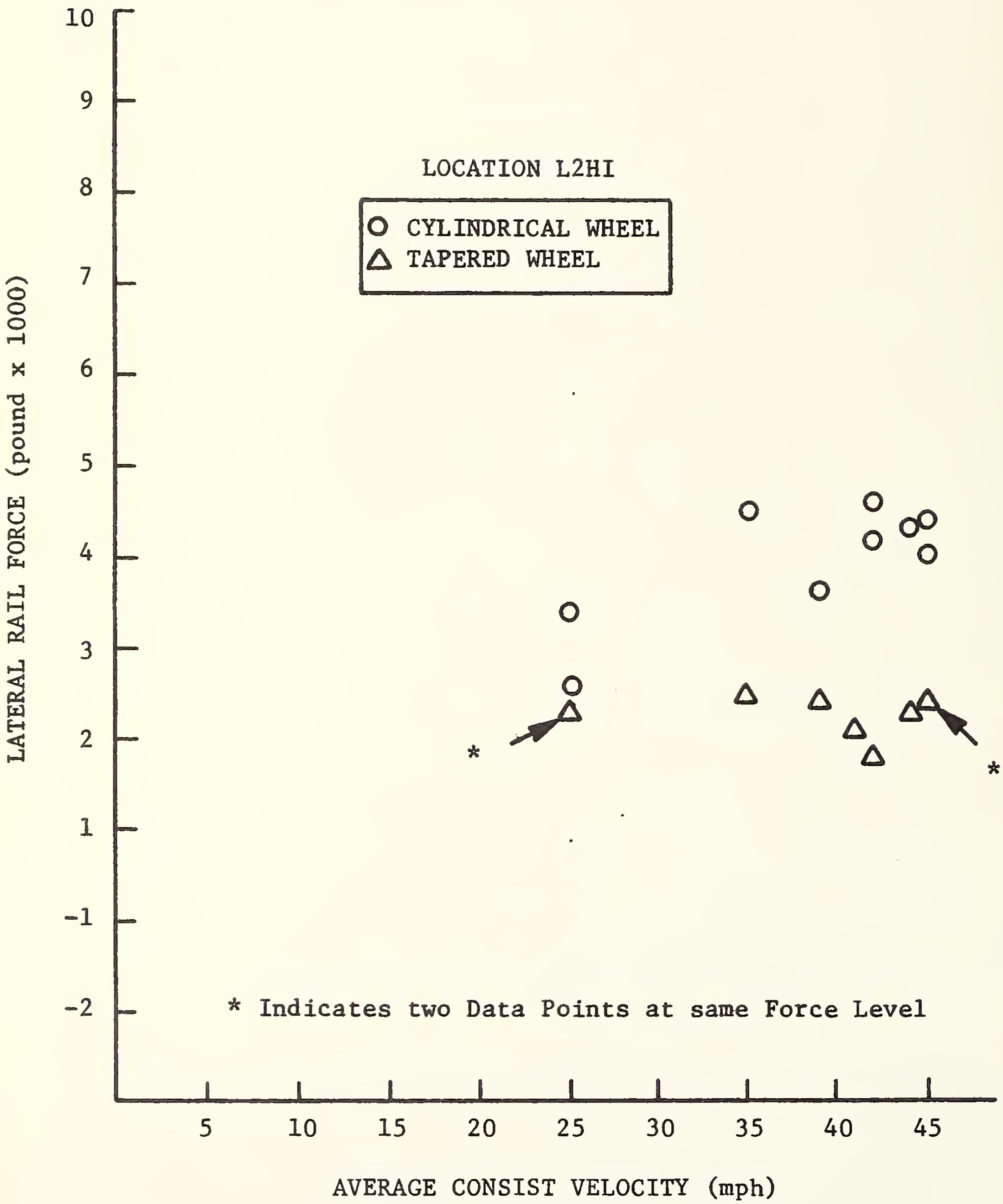


FIGURE 7. PLOT OF THE LATERAL RAIL FORCE PRODUCED AT CIRCUIT L2HI AT THE WASHINGTON NATIONAL AIRPORT SITE VS AVERAGE CONSIST VELOCITY FOR DRY RAIL WITH COAST CONDITION AND WMATA NORMAL GAGE

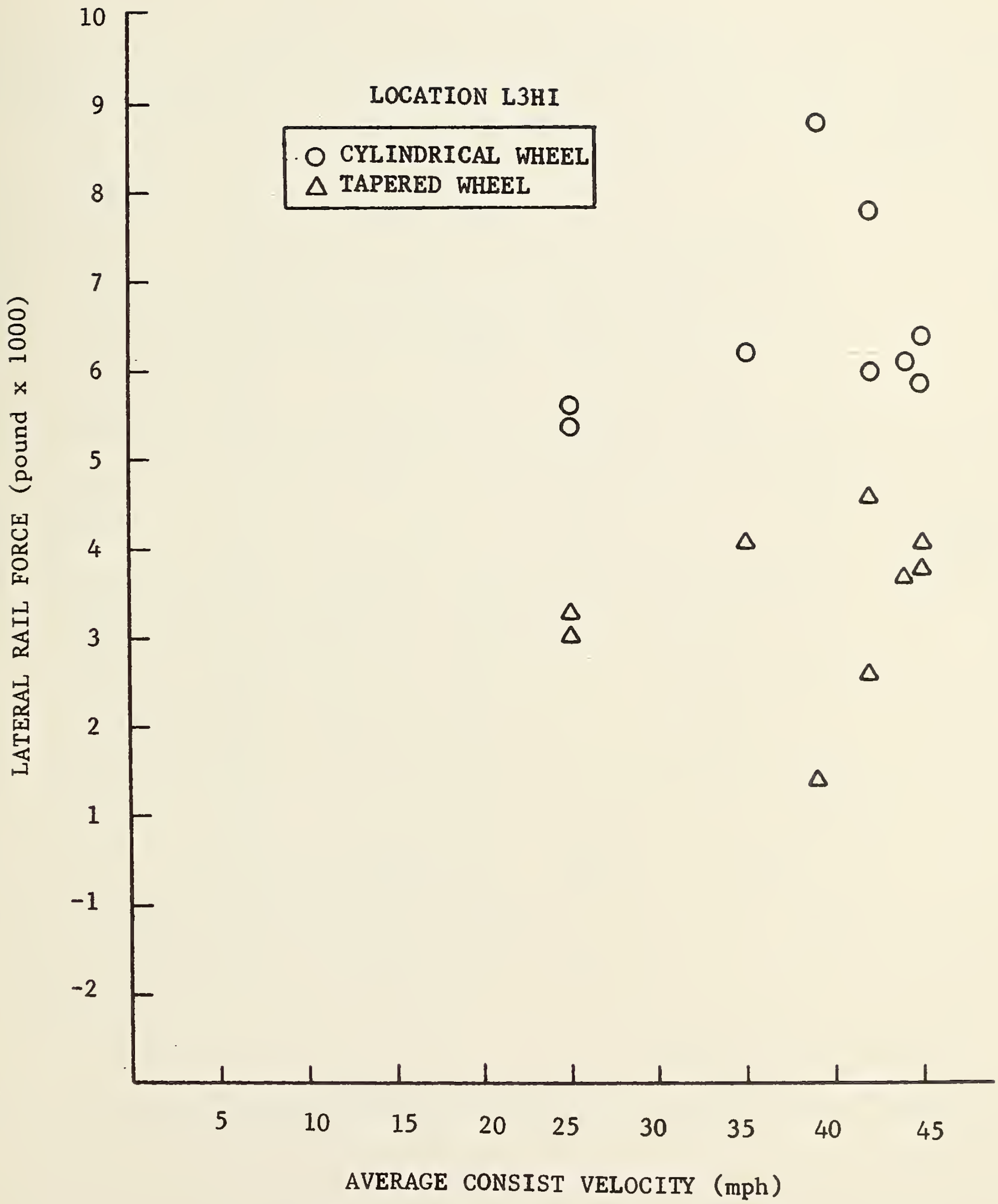


FIGURE 8. PLOT OF THE LATERAL RAIL FORCE PRODUCED AT CIRCUIT L3HI AT THE WASHINGTON NATIONAL AIRPORT SITE VS AVERAGE CONSIST VELOCITY FOR DRY RAIL WITH COAST CONDITION AND WMATA NORMAL GAGE.

TABLE 5. NATIONAL SITE, LEAD AXLES (1,5) HIGH RAIL, MEAN LATERAL W/R FORCE, DRY RAIL (FORCES IN POUND UNITS)

CONDITION	LOCATION L2	L3	L4	AVERAGE	SAMPLE SIZE
CYLINDRICAL WHEELS					
Accelerating	4540 (580)	5520 (700)	3990 (890)	4683 (734)	20
Coasting	4050 (450)	6460 (1110)	3850 (548)	4787 (760)	9
Braking	3670 (150)	4620 (730)	4250 (70)**	4180 (432)	4
				4550*(659)	
TAPERED WHEELS					
Accelerating	2730 (380)	4100 (670)	3140 (570)	3323 (553)	20
Coasting	2270 (210)	3400 (960)	2410 (879)	2693 (761)	9
Braking	1700 (330)	2450 (640)	2300 (0)**	2150 (509)	4
				2722*(617)	
REVENUE	3100	3800	3400	3433	

*Group Average

**Sample Size 2 for braking run at L4.

NOTE: The numbers in parenthesis are sample standard deviations with n-1 weighting.

TABLE 6. CALCULATED FLANGE FORCES, LEAD AXLES (1,5), HIGH RAIL, NATIONAL AIRPORT SITE, DRY RAIL (FORCES IN POUND UNITS)

CONDITION	LOCATION L2	L3	AVERAGE	SAMPLE SIZE
CYLINDRICAL WHEELS				
Accelerating	8910	10,930	9920	20
Coasting	8360	11,790	10,075	9
Braking	7360	8780	8070	4
			9355*	
TAPERED WHEELS				
Accelerating	6770	9000	7885	20
Coasting	5580	7290	6435	9
Braking	4190	5050	4620	4
			6313*	
REVENUE	6800	9000	7900	

*Group Average

for details of recording and data reduction system), it is not possible to calculate the flange force at location 4. The relative magnitudes of these flange forces are similar to that of the wheel rail forces shown in Table 5. On the average, for the coasting condition the tapered wheel profile results in flange forces 36 percent less than those associated with the cylindrical wheel profile. The revenue service runs with worn cylindrical wheels indicate forces about 21 percent less than those for the newly machined cylindrical wheels and 22 percent greater than those for the newly machined tapered wheels.

It is of interest to compare available theoretical results for W/R force and flange force with these experimental results. Weinstock and Greif[1] obtain closed form estimates of the forces for transit trucks from an analysis of rigid frame trucks in steady state curving. The wheel profiles are modeled as tapered wheel treads with vertical wheel flanges. The analysis includes the effect of flange friction, and force saturation leading to gross wheel sliding and nonlinear creep relationships. The theory predicts that for an 800-foot radius (the National Airport site), the trucks will be in a free curving orientation with flanging on the lead outer wheel and no flanging for the wheels of the trailing axle. Assuming a perfectly lubricated flange and a nonlinear creep condition with a W/R friction coefficient (adhesion level) of $\mu=.5$, Figures 8 and 9 of Reference 1 predict a W/R force at the outboard wheel of the lead axle of 9000 lbs and a flange force of 16,000. In the Appendix of Reference 1, the effects of flange friction are considered and an appropriate "knock-down" factor is listed as a function of flange/rail coefficient of friction. Assuming a value of 0.4 for this coefficient, the knock-down factor is 31 percent for the W/R force and 24 percent for the flange force for the linear creep case. Applying these factors to the aforementioned force levels, leads to a predicted W/R force of 6200 lbs and a flange force of 12,100 lbs. From Tables 5 and 6, the coasting condition W/R force recorded at L3 from the test is 6460 lbs and the mean flange force is 11,790 lbs. This proves to be an excellent comparison to the theoretical model for the experimental case involving cylindrical wheels. The forces from the tests for the tapered wheels are much less than the forces for the cylindrical wheels (W/R = 3400 lbs, Flange = 7290 lbs at L3) and consequently, these results do not compare as well with those in the theoretical model. It is of interest to examine how taper angle affects truck performance. For a rigid truck, Reference 1 shows that a taper angle of 1/20 will reduce the force results by less than 8 percent for a transit truck at a typical curve such as that at the National site. However, the effect of taper is much more pronounced for a flexible truck design, such as a transit truck. In Reference 3, Newland studied the minimum radius curve that can be negotiated by the flexible truck without wheel slip,

$$R \leq \frac{\frac{2f_T h\ell}{\mu W}\left[1 + h^2 \frac{f_L}{f_T}\left\{1 + \frac{\alpha\ell}{r_o}\left(\frac{f_T}{K_y h\ell}\right)\right\}^2\right]^{1/2}}{\left(1 + \frac{\alpha\ell}{r_o}\frac{f_L f_T}{K_y K_\psi}\right)}$$

where, as listed in Reference 1, the symbols represent,

$f_L,\ f_T$ = creep coefficient (lb) in lateral, tangential direction

ℓ = half of track gage

h = ratio of wheel base to track gage

μ = coefficient of friction

W = wheel load

α = wheel conicity

r_o = wheel radius of undisplaced wheelset

K_y = effective lateral stiffness of flexible truck (lb/inch)

K_ψ = effective yaw stiffness of flexible truck (in-lb/rad)

This formula shows that the flexibility of the truck can only be utilized through the wheel taper α. Modeling the flexible transit truck with cylindrical wheels (α=0) as a rigid truck is a reasonable assumption which leads to accurate force comparisons. Modeling the flexible transit truck with tapered wheels as a rigid truck is not as realistic a condition and the force level comparisons are not as good. Further research is now being done to define the influence of truck flexibility and wheel conicity on force levels. It should be noted in making comparisons of the analytical and experimental results that the effective conicity of the tapered wheels may be difficult to define precisely. For example, the tapered wheels used in the present tests were a British Rail profile which has two effective conicities determined from calculations of the rolling radius difference as a function of wheelset displacement. This profile shows two effective conicities - conicity of 0.04 to 0.05 in the tread region and an effective conicity of 0.11 for the 56.5 inch gage as the flange is approached. The 56.25 inch gage would make the effective conicity about 0.15. Wear of the rail may have had a further influence in increasing the effective conicity.

b) <u>Accelerating and Braking Runs</u>

Two acceleration rates were tested: 1.5 mph/second and 3.0 mph/second. Similarly, two braking rates were tested: normal service braking and maximum braking. The trends from both these conditions were similar to the results from those obtained from coasting. Namely, that during these runs, maximum load levels occurred at the L3 location and the cylindrical wheel loads were greater than the tapered wheel loads. It is difficult to ascertain any trend for the accelerating run when plotting the data against average velocity as shown in Figure 9, although the results do appear to have more variance in comparison to the coasting data of Figure 7. A comparison of W/R loads for the lead axles (1,5) in the accelerating, coasting and braking modes is shown in Table 5. On the average, the accelerating runs produce higher forces than the braking runs. Although, this is not true at location L4, it is worth noting that the L4 braking runs are based on only two samples. For cylindrical wheels, the average lateral forces for the three modes are within 14 percent of each other. For tapered wheels, the braking forces are 35 percent less than the forces for the accelerating runs. For the accelerating modes, the tapered wheel W/R

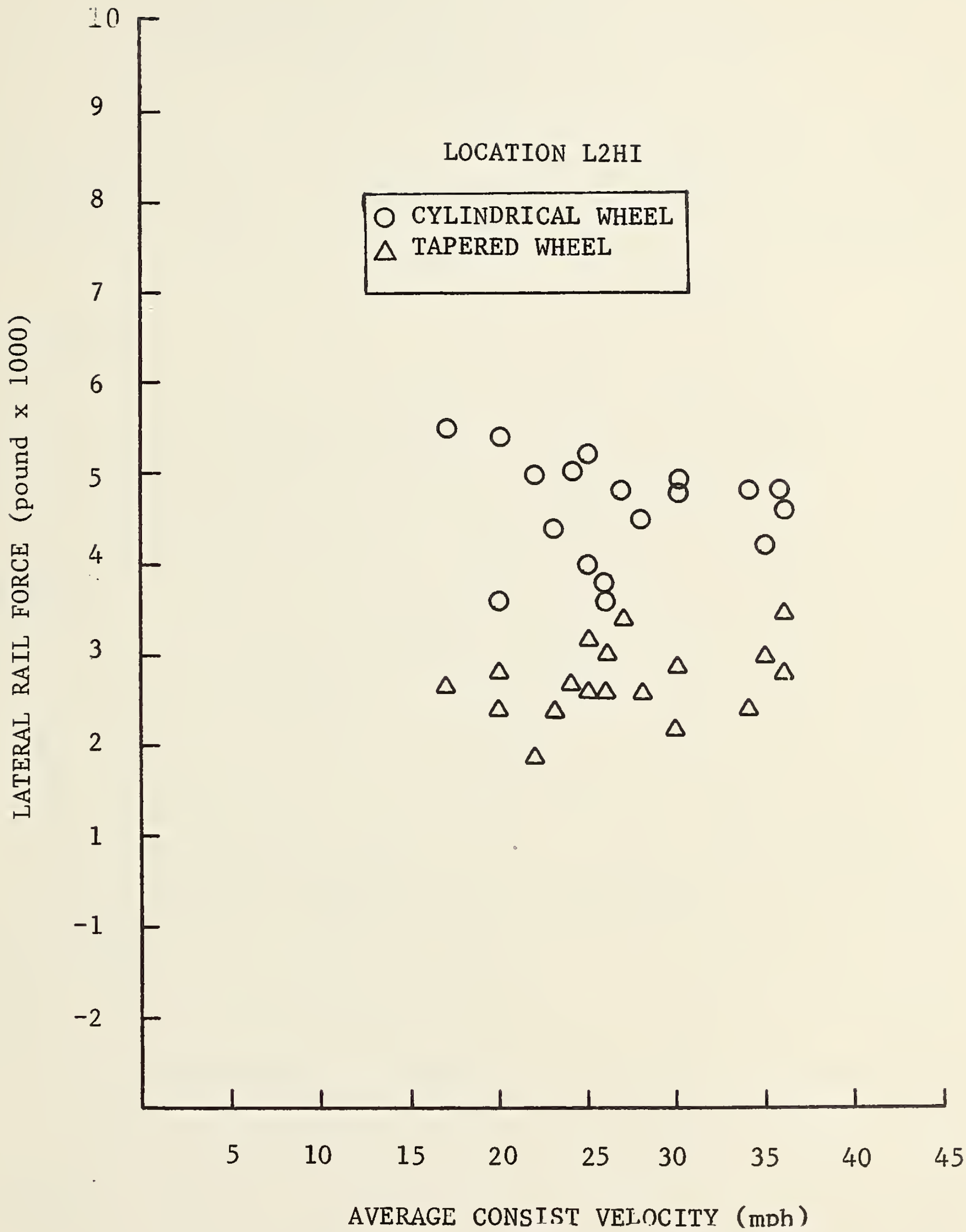


FIGURE 9. PLOT OF THE LATERAL RAIL FORCE PRODUCED AT CIRCUIT L2HI AT THE WASHINGTON NATIONAL AIRPORT SITE VS AVERAGE CONSIST VELOCITY FOR DRY RAIL WITH ACCELERATING CONDITION AND WMATA TIGHT GAGE

force is 29 percent less than that for the cylindrical wheel. For the braking mode, the tapered wheel W/R force is 49 percent less than that for the cylindrical wheel. The flange force trend shown in Table 6 is similar to that for the W/R forces. The maximum values occur at location L3 and the tapered wheels significantly lower the force levels. For example, in the braking mode the tapered wheel flange force is 43 percent less than the flange force for the cylindrical wheel.

2.2.2 Lubricated Rails

Some runs were made with lubricant applied to the rails. There were several variations in the testing conditions, with tests involving lubrication of the high rail only, the low rail only, and both the high and low rail. Because of difficulties in controlling the application of lubricant and keeping it in place as the test consist traversed the curve, it is somewhat difficult to interpret the data for each of the foregoing categories. W/R and flange forces of the runs conducted on the 18th and 19th of August with lubrication on the high rail are tabulated in Table 7. To fully assess the effect of rail lubrication, these results should be compared to the dry rail results in Tables 5 and 6. On the average, these lateral forces acting on the lubricated rails are <u>higher</u> than the corresponding forces on dry rail. The average coasting cylindrical W/R force is 23 percent higher when the rail is lubricated and the corresponding increases for the tapered wheel is 57 percent. The highest forces occurred at the L3 location. The maximum mean lateral W/R force for the cylindrical wheel is about the same (6400 lbs) for both lubricated and dry rail. The maximum mean lateral W/R force for the tapered wheel increased by 26 percent due to lubrication. Flange forces also increased due to the lubrication of the rail. Although the flange forces increase, it should be noted that the <u>flange friction force</u> is reduced since the coefficient of friction between wheel and rail is reduced by the lubrication. This reduction in flange friction force may have a significant impact on reducing wear conditions. Another effect shown in Table 7 from comparison of results for tapered and cylindrical wheels is that the tapering of the wheel is not as effective in reducing the flange force under lubricated conditions as under dry conditions.

TABLE 7. LUBRICATED HIGH RAIL

NATIONAL SITE, LEAD AXLES (1,5) HIGH RAIL
LATERAL W/R FORCE AND FLANGE FORCE
(FORCES IN POUND UNITS)

a) WHEEL/RAIL FORCES

CONDITION	LOCATION				
	L2	L3	L4	AVERAGE	SAMPLE SIZE
CYLINDRICAL WHEELS					
Accelerating	5220	6300	5100*	5540	4
Coasting	6100	6400	5200	5900	3
Braking	3550	5750	3950	4417	2
TAPERED WHEELS					
Accelerating	3450	4870	4400*	4240	4
Coasting	3530	5160	3960	4217	3
Braking	2750	4250	2250	3083	2

b) FLANGE FORCES

CYLINDRICAL WHEELS					
Accelerating	8900	11,040	-	9970	4
Coasting	10,960	12,100	-	11,500	3
Braking	7730	10,890	-	9310	2
TAPERED WHEELS					
Accelerating	8350	11,030	-	9690	4
Coasting	8270	10,900	-	9585	3
Braking	6880	8570	-	7725	2

*Data from only one run.

Some of these force results for lubricated rail may be partially explained on the basis of theoretical models. The presence of lubrication on the tread will diminish the creep forces developed during the W/R interaction, which tends to lower both the W/R force and the flange force. On the other hand, as shown in the Appendix of Reference 1, flange lubrication will increase both flange force and W/R force. The calculations show that changing from a flange coefficient of friction of 0.4 to zero (going from flange friction to perfect lubrication), the flange force on the lead outer wheel for the free curving region increases by 31 percent and the W/R force increases by 43 percent. It may well be that this flange lubrication effect is dominating the tread lubrication effect leading to the increase in lateral forces shown in Table 7.

2.2.3 Wide Gage

A series of wide gage runs were made on August 22 at the National Airport Test site. A full description of the track gage and test series is given in Tables A-1 and B-7 of Volume II. The results of this series of runs are presented in Table 8. Since only the accelerating mode was tabulated and analyzed, these results are compared to the accelerating mode for the normal gage runs from Table 5. Widening the gage tends to lower both the W/R forces and the flange forces with respect to levels attained with normal gage. The maximum cylindrical W/R force 5520 lbs is reduced 45 percent to 3060 lbs and the maximum tapered W/R force of 4100 lbs is reduced 34 percent to 2720 lbs, by the wide gage condition. The maximum cylindrical flange force for the accelerating mode 10,930 lbs is reduced 38 percent (to 6730 lbs) and the comparable tapered wheel reduction is 37 percent, due to the wide gage condition. In terms of overall curve averages, widening the gage reduces the flange force for the cylindrical wheel by 33 percent and for the tapered wheel by 31 percent.

Tests were also run on August 22 for wide gage with lubrication on the high rail and subsequent tests with lubrication on the low rail. A comparison of the wide gage results for dry rail and lubed high rail is presented in Table 9. The effect of lubrication is to increase both the W/R forces and flange forces. When the high rail is lubricated the average cylindrical W/R force is increased by 30 percent over the wide gage dry rail force while

TABLE 8. COMPARISON OF WIDE GAGE AND NORMAL GAGE RUNS, DRY RAIL NATIONAL SITE, LEAD AXLES, (1,5), HIGH RAIL ACCELERATING CONDITIONS (FORCES IN POUND UNITS)

a) WHEEL/RAIL FORCES

GAGE	WHEEL	LOCATION L1	LOCATION L2	LOCATION L3	AVERAGE	SAMPLE SIZE
Wide	Cylindrical	2400	3060	1900	2453	5
Normal*	Cylindrical	4540	5520	3990	4683	20
Wide	Tapered	2060	2720	1580	2120	5
Normal*	Tapered	2730	4100	3140	3323	20

*From Table 5.

b) FLANGE FORCES

GAGE	WHEEL	LOCATION L1	LOCATION L2	LOCATION L3	AVERAGE	SAMPLE SIZE
Wide	Cylindrical	5850	6730	-	6290	5
Normal*	Cylindrical	8910	10,930	-	9920	20
Wide	Tapered	5260	5650	-	5455	5
Normal*	Tapered	6770	9000	-	7885	20

*From Table 6.

TABLE 9. WIDE GAGE RUNS, NATIONAL SITE
LEAD AXLES (1,5), HIGH RAIL
ACCELERATING CONDITIONS
(FORCES IN POUND UNITS)

a) WHEEL/RAIL FORCES

CONDITION	LOCATION				
	L2	L3	L4	AVERAGE	SAMPLE SIZE
DRY RAIL (TABLE 8)					
Cylindrical	2400	3060	1900	2453	5
Tapered	2060	2720	1580	2120	5
LUBED RAIL (HI)					
Cylindrical	3120	4020	2480	3207	5
Tapered	2080	3080	1760	2307	5

b) FLANGE FORCES

DRY RAIL (TABLE 8)					
Cylindrical	5850	6730	-	6290	5
Tapered	5260	5650	-	5455	5
LUBED RAIL (HI)					
Cylindrical	7130	8490	-	7810	5
Tapered	5540	6560	-	6050	5

the average tapered-wheel force is increased by 9 percent; the average cylindrical flange force is increased by 24 percent while the average tapered flange force is increased by 11 percent.

3.0 CONCLUSIONS AND RECOMMENDATIONS

3.1 CONCLUSIONS

Measurements of wheel/rail forces were made with strain gages mounted on the high rail and the low rail of the Washington Metrorail at an 800 ft radius curve at Washington National Airport and at a 1500 ft radius curve opposite the Brentwood Yard. Lateral and vertical forces were measured for revenue consists with worn* cylindrical wheels and for a 2-car test consist, one car equipped with unworn cylindrical wheels and the other equipped with unworn 1:20 tapered wheels. Flange forces were calculated from the measured values of the wheel/rail forces (see the Appendix).

This section concentrates on the mean flange force between the outer wheel and high rail of the lead axle, because this force is one of the prime causes for wheel/rail wear. This average flange force at the Washington National Airport is as follows:

For tight gage (dry rail, Table 6):

9400 pounds, unworn cylindrical profile

6300 pounds, unworn tapered profile

7900 pounds, worn (revenue) cylindrical profile

For widened gage (dry rail, Table 8):

6300 pounds, unworn cylindrical profile

5500 pounds, unworn tapered profile

The following observations are made:

a) Going from an unworn cylindrical wheel to an unworn tapered wheel reduces the average flange force by 33 percent. Going from a worn (revenue) cylindrical wheel to an unworn tapered wheel reduces the average flange force by 20 percent.

b) Widening the gage reduces the unworn cylindrical wheel flange force by 33 percent. Widening the gage and going from an unworn cylindrical wheel to an unworn tapered wheel reduces the flange force a total of 41 percent.

*"Worn" refers to the complete range of revenue wheel profiles.

c) Although a review of the literature reveals no generally accepted relationship between flange force and wheel/rail wear, flange force reductions of 20 percent to 40 percent are expected to result in a significant reduction in wheel/ rail wear.

d) The flange force value at Brentwood is 1600 pounds for worn (revenue) cylindrical wheels. This value is 80 percent less than the corresponding 7900 pound flange force for revenue runs at Washington National Airport. This difference is accounted for by curve radius twice that at National and the presence of rain as a lubricant during the Brentwood test runs.

e) The mean flange force value for worn (revenue) cylindrical wheels, (see Table 1), at the point of maximum wear at the Washington National Airport (L3, 9000 lbs) is 32 percent greater than the mean value approximately 78 inches away (L2, 6800 lbs). This difference combined with measured truck yaw motion (.3° in 1.3°) that is highly repetitive, indicates definite dynamic yaw activity as the truck negotiates the Washington National Airport curve.

f) Because of difficulties with controlling lubrication during the tests, both the gage side and the tread of the rail were lubricated. As a result, lubrication was observed either to decrease or to increase wheel/rail forces. Because the advantage of lubrication is to reduce adhesion and wear independent of lateral force, no comments on the effectiveness of lubrication can be made.

g) There was no pronounced pattern relating forces to velocity, accelerating, coasting, or braking conditions. The absence of a velocity relationship indicates that the existence of an unbalanced speed on the Washington National Curve is not critical within the specified operating speed range.

3.2 RECOMMENDATIONS

In consideration of the significant reduction in flange forces observed as a result of using tapered wheels and widening the gage, it is recommended that:

a) On a progressive basis the cylindrical wheels should be replaced by tapered wheels, carefully monitoring the results of their use.

b) The tight gage on curves with high wear be widened to standard gage at the time of rail replacement or reversal.

c) An ongoing measurement program be established to determine the effect on wheel and rail wear of (a) and (b) above.

Wear studies should be conducted at WMATA, MARTA, and other properties to provide baseline data for determining the existence of excessive wear, the causes of wear, and types of wear related to transit use.

Measurements of truck characteristics and analytic studies should be performed as part of a longer term program to determine optimum curving vs speed tradeoffs for conventional trucks. The feasibility of implementing the use of radial trucks should be investigated.

Lubrication and restraining rail should be implemented on curves with chronic wear consistent with the practice on other transit properties.

REFERENCES

1. Weinstock, H. and R. Greif, "Analysis of Wheel Rail Force and Flange Force, During Steady State Curving of Rigid Trucks," TSC Working Paper No. WP-743-C-13-81, DOT/Transportation Systems Center, Cambridge MA, January 1980.

2. Newland, D.E., "Steering a Flexible Railway Truck on Curved Track," Journal of Engineering for Industry, August 1969.

APPENDIX

ESTIMATING FLANGE FORCES

A.1 FLANGE FORCES

The vertical and lateral wheel/rail forces are recorded at preselected positions on the rails. The calculation of the flange force is then performed as follows. Consider a wheelset which is flanging on the <u>high</u> rail as shown in Figure A-1. The forces shown are acting on the wheelset (the forces on the rail act in the opposite direction) with

F_c = creep force

F_f = flange force

v_c = creep velocity

f = creep coefficient

V = vertical load

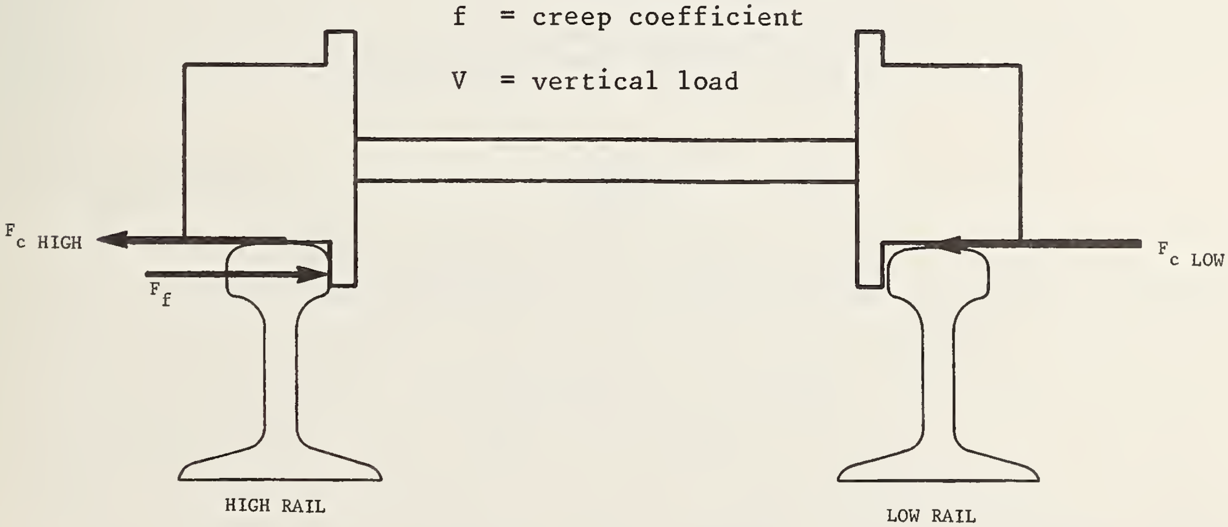


FIGURE A-1. DEFINITION OF WHEELSET LATERAL FORCE (FLANGING ON HIGH RAIL, FORCES ACTING ON WHEELSET)

Since the axle is rigid, the lateral creep velocity (i.e. the difference between the actual lateral velocity and the pure rolling velocity) is the same for each wheel,

$$v_{c_H} = v_{c_L} \tag{1}$$

The creep force on each wheel is proportional to the respective creep velocity,

$$F_{C_H} = f_H v_{C_H}$$

$$= f_H v_{C_L}$$

$$= \frac{f_H}{f_L} F_{C_L} . \qquad (2)$$

The difference in the vertical wheel load between high and low rail will affect the lateral creep coefficients. For a first order analysis the variation in creep coefficient with vertical load may be assumed in the form

$$\frac{f_H}{f_L} = \left(\frac{V_H}{V_L}\right)^{2/3} . \qquad (3)$$

The creep force on the high rail is then found from (2) as

$$F_{C_H} = F_{C_L} \left(\frac{V_H}{V_L}\right)^{2/3} . \qquad (4)$$

The resultant wheel/rail force measured at the site is then

$$F_{W/R\ High} = F_f - F_{C_H} \qquad (5)$$

leading to the flange force relation

$$F_f = F_{W/R\ High} + F_{C_L} \left(\frac{V_H}{V_L}\right)^{2/3} . \qquad (6)$$

A similar analysis for a flanging condition on the low rail leads to the flange force relationship

$$F_f = F_{W/R_{Low}} + F_{C_H} \left(\frac{V_L}{V_H}\right)^{2/3} \qquad (7)$$

As an example of the use of this flange force formula, consider run 17-2 (40 mph coast) for a truck with cylindrical wheels as shown in Figure A-2. Positive forces tend to push rails outward from the centerline.

It is assumed that both axles are flanging on the outer rail. It then follows that

$$F_{f\ Lead} = 6000 + 3500 \left(\frac{17.4}{7.6}\right)^{2/3} \approx 12{,}000 \text{ lbs}$$

$$F_{f\ Trail} = 400 + 700 \left(\frac{14.6}{9.6}\right)^{2/3} \approx 1300 \text{ lbs} \tag{8}$$

In these calculations, an assumption must be made concerning flanging on the high or low rail so that either equation (6) or (7) can be used. For both of the axles shown in Figure A-2, the vertical wheel load is much greater on the high rail than on the low rail and it is reasonable to assume that the flange forces accordingly also act on the high rail.

A.2 TRUCK FORCES

There is evidence of substantial dynamic activity in the trucks as the site is traversed. The analysis of the truck rotation data will be useful in understanding this effect. As an insight into this phenomenon, consider the truck of run 17-2 with cylindrical wheels traversing the site and positioned at locations 2-3 and 3-4. The illustration of location 2-3 is identical to Figure A-2 and is reproduced as A in Figure A-3, while location 3-4 is B.

As calculated on Figure A-3, the net moment acting on the truck changes dramatically from position A to position B. In addition, the orientation of this moment changes, which could imply significant yaw activity between the two positions. One possible scenario is that while the lead axle is flanging on the high rail during motion through the test site, the trailing axle is "fish-tailing" and alternating on its flanging position. This alternating motion would imply that pure steady-state curving behavior is not being maintained as the truck moves along the curved site.

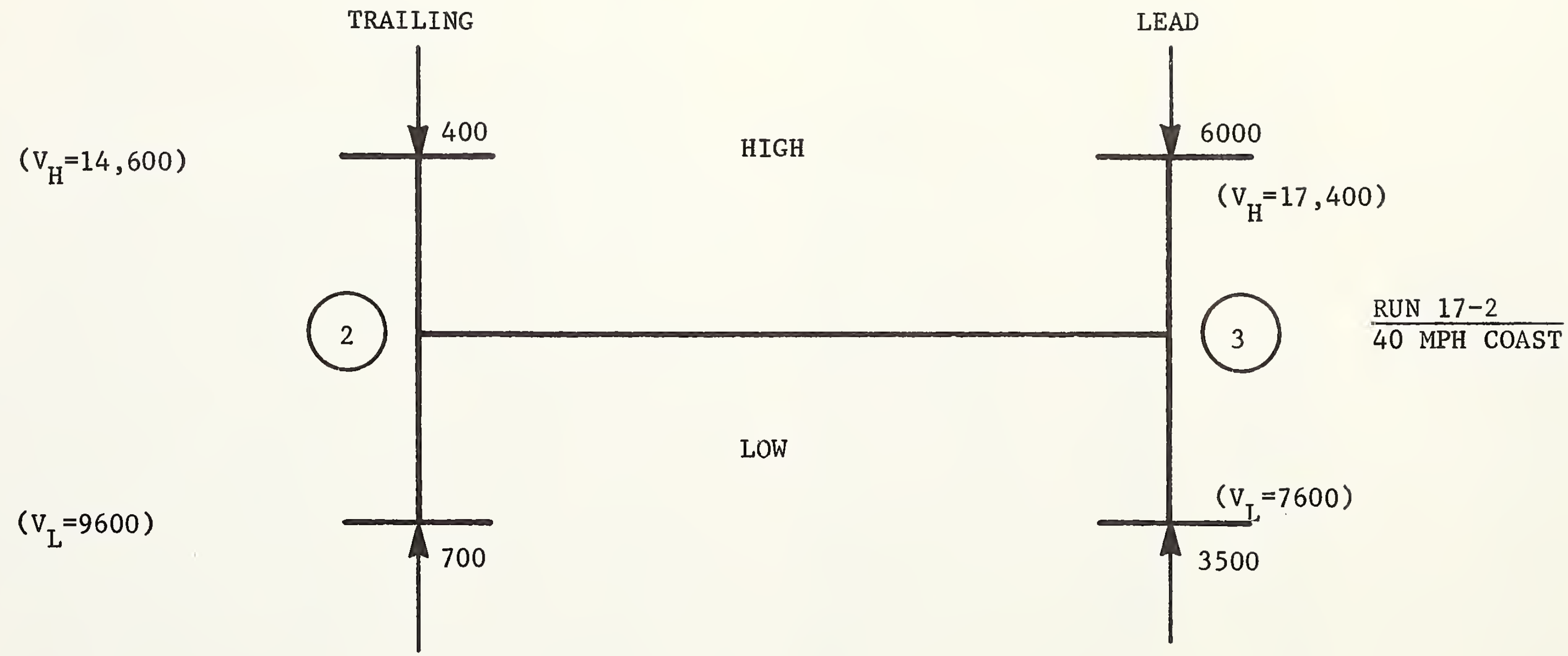
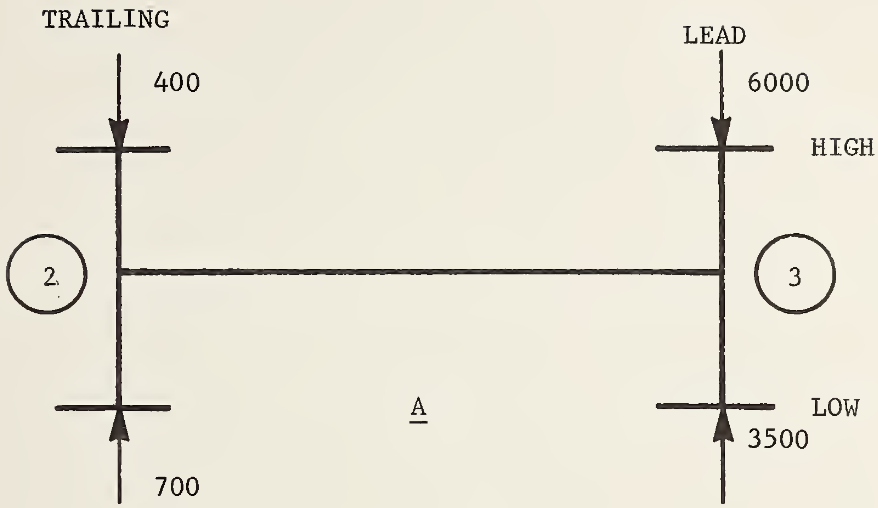


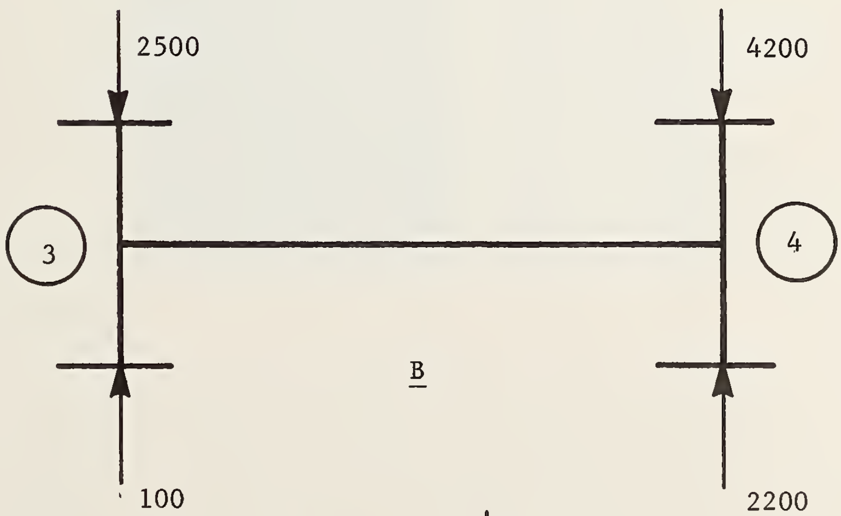
FIGURE A-2. WHEEL/RAIL FORCES ON LEAD TRUCK
(CYLINDRICAL WHEELS, LOCATIONS 2 AND 3)



NET FORCE = 2200 lbs

NET MOMENT = (2500 + 300) 43.5

$= 12.2 \times 10^{4}$ in - lbs



NET FORCE = 4400 lbs

NET MOMENT = (2000 - 2400) 43.5

$= - 1.74 \times 10^{4}$ in - lbs

FIGURE A-3. NET LEAD TRUCK FORCE AND MOMENT (RUN 17-2, CYLINDRICAL WHEELS, 40 MPH COAST)

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